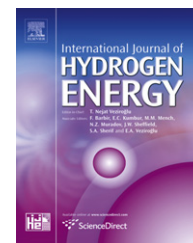


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Twin cylinder alpha stirling engine combined model and prototype redesign

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ABSTRACT

This article shows the development of a twin cylinder alpha type Stirling engine model, as well as the redesign of a prototype, its manufacture and preliminary results. The mechanical model is coupled with a thermodynamic model which allows an analysis and comparison of the theoretical and experimental behavior of the machine and a better understanding of the whole process. On the other side, a prototype redesign is presented. Thermal circuit was improved in order to eliminate losses, reduce dead volumes and improve heat transfer to the working fluid. Modifications of the seal system and the pistons design are included as well. An equation for the sizing of the flywheel is developed and finally the results obtained in tests between the new and the previous design are compared.

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1. Introduction

1.1. Stirling cycle machines. Design and simulation methods

Stirling cycle machines can be used as engines, generators, combined heat and power equipment (CHP), cooling machines or heat pumps. In these devices, Stirling cycle has the advantages of allowing the use of alternative energy sources and avoiding environmentally harmful refrigerants.

Many simulation models for Stirling machines can be found in literature. However, most of them study the

thermodynamic problem, without linking it to the mechanic one [1–3,9]. Moreover, Chin-Hsiang Cheng [4] has recently presented a model that takes into account this connection, which shows in a complete analysis the performance of the device. Saravia et al. [5] also present a model including mechanical variables. The present work uses the same global approach for the development of the model as the one in Cheng [4], but developed for a twin cylinder V-90 alpha type engine. This paper presents also a redesign and the first results of an engine built with the methodology and design tools shown in [6,7]. Some mechanical aspects of importance and the adopted solutions are mentioned.

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Nomenclature			
<i>Variables and measure units</i>		Wo	Work, J
α	Angular acceleration, rad/s ²	x	Cylinder axis direction
β	Angle between a rod and its piston, rad	x _p	Piston position, m
θ	Angle between the crankshaft and the expansion cylinder axis, rad	y	Perpendicular cylinder axis direction
τ	Time, s	<i>Subscripts</i>	
ω	Angular velocity, rad/s	1	Expansion piston
a _p	Linear piston acceleration, m/s ²	2	Compression piston
dt	Time step, s	3	Crankshaft-rod joint
F	Forces applied in the expansion piston and rod, N	4	Compression rod-piston joint
g	Gravitational acceleration, 9.81 m/s ²	4–3	Compression piston rod
G	Forces applied in the compression piston and rod, N	5	Expansion rod-piston joint
I	Moment of inertia, kg m ²	5–3	Expansion piston rod
L	Rod length, m	c	Compression
L _p	Piston length, m	c2	Compression cylinder
m	Mass, kg	ca	Crankcase
N	Power, W	ci	Initial load
P	Pressure, Pa	cig	Crankshaft
Q	Heat exchange, J	cil	Cylinder
R	Radius, m	e	Expansion
R _{gas}	Gas constant (Helium), J K/kg	e1	Expansion cylinder
T	Torque, N·m	f	Friction
t	Temperature, K	G	Rod center of mass
T _c	Cold source temperature, K	h	Heater
T _h	Hot source temperature, K	k	Cooler
V	Volume, m ³	n	Rod axis direction
v _p	Linear piston velocity, m/s	p	Piston
W	Weight, N	q	Load
		r	Regenerator
		t	Perpendicular to rod axis direction
		tan	Tangential direction

1.2. Stirling engine, heat pumps and refrigeration

The engine mode of operation of the machine is more versatile than other engines, since the heat source is external, and offers the advantage of adapting to various fuel types and alternative energy sources such as wood pellets, hydrogen and biogas for example.

As heat pumps, these machines are capable of transferring heat from a low-temperature source to higher temperature one through the input of mechanical work. The commercially existing air-conditioning equipments use this principle, but the disadvantage of their use is that they need refrigerants such as HCFCs (R-22), HFCs and others. Here, Stirling cycle heat pumps can have a major advantage since they use air, nitrogen or helium as working fluids, all of them without prejudice to the environment. In that sense, it is of great interest the applicability of the Stirling cycle to geothermal heat pumps, which absorb heat from a buried coil and deliver it inside a room, achieving a coefficient performance (COP) with values between 2 and 5. The applications of Stirling cycle as cooling machines are varied and become important especially at low temperatures (below –20 °C) where the traditional cycles have low performance.

2. Model

2.1. Thermodynamic sub-model

A simplified isothermal type thermodynamic model is used here, mainly as a support for the mechanic model in the next section. In future versions of the program, the model will include the corresponding losses of heat and pressure.

To calculate the mass present in each machine part, initial load pressure is used:

$$\begin{aligned} m_c &= P_{ci} \cdot V_{c2} / (R_{gas} \cdot t_c) \\ m_k &= P_{ci} \cdot V_k / (R_{gas} \cdot t_k) \\ m_r &= P_{ci} \cdot V_r / (R_{gas} \cdot t_r) \\ m_h &= P_{ci} \cdot V_h / (R_{gas} \cdot t_h) \\ m_e &= P_{ci} \cdot V_{e1} / (R_{gas} \cdot t_e) \end{aligned} \quad (1)$$

From mass conservation, the total mass is the sum of all them at each moment:

$$m = \frac{P}{R_{gas}} \left(\frac{V_{c2}}{t_c} + \frac{V_{e1}}{t_e} + \frac{V_k}{t_k} + \frac{V_r}{t_r} + \frac{V_h}{t_h} \right) \quad (2)$$

Writing it for P gives,

$$P = \frac{m \cdot R_{gas}}{\left(\frac{V_c}{t_c} + \frac{V_e}{t_e} + \frac{V_k}{t_k} + \frac{V_r}{t_r} + \frac{V_h}{t_h} \right)} \quad (3)$$

as a simplification, uniform pressure is considered, $P = P_{e1} = P_{c2}$.

The total volume is calculated using:

$$V = V_{c2} + V_{e1} + V_r + V_k + V_h \quad (4)$$

Using P , the mechanical model will calculate the torque and the resulting angular velocity at each step and in consequence the volume variation ΔV . Then, in case of compression:

$$Q_k = -W_{oc} = P \cdot \Delta V \quad (5)$$

And during expansion:

$$Q_h = W_{oe} = P \cdot \Delta V \quad (6)$$

2.2. Dynamic sub-model

A schematic diagram of an alpha type engine is illustrated in Fig. 1. Pistons have a phase angle of $\pi/2$. The assembly of the exchangers is also shown. Rods join the crankshaft at a radius R_{cig} and an angle θ from the expansion cylinder axis.

For this first version of the model, ideal exchangers are supposed, so the heater wall temperature and that of the expansion cylinder are the same and equal to T_h . The same applies to the cooler and the compression cylinder with T_c . Two more considerations are taken into account: the rod center of gravity is on its half length, and all parts are strictly rigid. Power is delivered in the main shaft and will be obtained from the forces over pistons and rods.

The equation for the expansion piston position is:

$$x_{p1} = L_p + L \cos(\beta_1) + R_{cig} \cos \theta \quad (7)$$

and for the compression one:

$$x_{p2} = L_p + L \cos(\beta_2) + R_{cig} \sin \theta \quad (8)$$

where

$$\beta_1 = \arcsin\left(\frac{R_{cig}}{L} \sin \theta\right) \quad (9)$$

and

$$\beta_2 = \arcsin\left(\frac{R_{cig}}{L} \cos \theta\right) \quad (10)$$

The volume of the expansion cylinder is:

$$V_{e1} = [L(1 - \cos \beta_1) + R_{cig}(1 - \cos \theta)] \pi R_{e1}^2 \quad (11)$$

and the volume in the compression cylinder:

$$V_{c2} = [L(1 - \cos \beta_2) + R_{cig}(1 - \sin \theta)] \pi R_{e2}^2 \quad (12)$$

Differentiating equations (7) and (8) with respect to time, the linear velocity of both pistons is calculated,

$$v_{p1} = -L\omega_{5-3} \sin \beta_1 - R_{cig} \omega \sin \theta \quad (13)$$

$$v_{p2} = -L\omega_{4-3} \sin \beta_2 + R_{cig} \omega \cos \theta \quad (14)$$

Where:

$$\omega_{5-3} = \frac{d\beta_1}{dt} = \frac{\omega R_{cig} \cos \theta}{L \cos \beta_1} \quad (15)$$

$$\omega_{4-3} = \frac{d\beta_2}{dt} = -\frac{\omega R_{cig} \sin \theta}{L \cos \beta_2} \quad (16)$$

In these equations ω is the angular velocity of the flywheel (derivative of θ with respect to time).

To obtain accelerations, equations (13) and (14) are differentiated:

$$a_{p1} = -L\omega_{5-3}^2 \cos \beta_1 - L\alpha_{5-3} \sin \beta_1 - R_{cig} \omega^2 \cos \theta - R_{cig} \alpha \sin \theta \quad (17)$$

$$a_{p2} = -L\omega_{4-3}^2 \cos \beta_2 - L\alpha_{4-3} \sin \beta_2 - R_{cig} \omega^2 \sin \theta + R_{cig} \alpha \cos \theta \quad (18)$$

where

$$\alpha_{5-3} = \frac{R_{cig}}{L \cos \beta_1} (\alpha \cos \theta - \omega^2 \sin \theta + \omega \omega_{5-3} \tan \beta_1 \cos \theta) \quad (19)$$

$$\alpha_{4-3} = \frac{R_{cig}}{L \cos \beta_2} (-\alpha \sin \theta - \omega^2 \cos \theta + \omega \omega_{4-3} \tan \beta_2 \sin \theta) \quad (20)$$

and α is $d\omega/dt$.

Torques and forces over the expansion rod and piston are analyzed now, following Fig. 2.

Two convenient reference system are used, the x-y fixed to the piston (translation) and the n-t fixed to the rod (rotation-translation).

Torque and force equilibrium equations from Fig. 2 are:

$$\begin{aligned} \sum T_3 &= -T_{p1} + \frac{L}{2} F_{Gt} + L F_{5y} \sin \beta_1 + L F_{5x} \cos \beta_1 \\ &\quad - \frac{L}{2} W_1 \sin\left(\frac{\pi}{4} + \beta_1\right) = 0 \end{aligned} \quad (21)$$

in n_1 direction,

$$\sum F_n = -F_{Gn} + F_{3n} - F_{5y} \cos \beta_1 + F_{5x} \sin \beta_1 + W_1 \cos\left(\frac{\pi}{4} + \beta_1\right) = 0 \quad (22)$$

and in t_1 direction,

$$\sum F_t = F_{Gt} + F_{3t} + F_{5y} \sin \beta_1 + F_{5x} \cos \beta_1 - W_1 \sin\left(\frac{\pi}{4} + \beta_1\right) = 0 \quad (23)$$

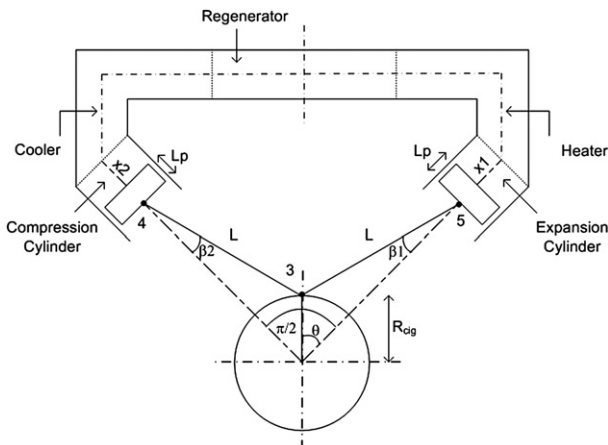


Fig. 1 – Schematic alpha type Stirling engine.

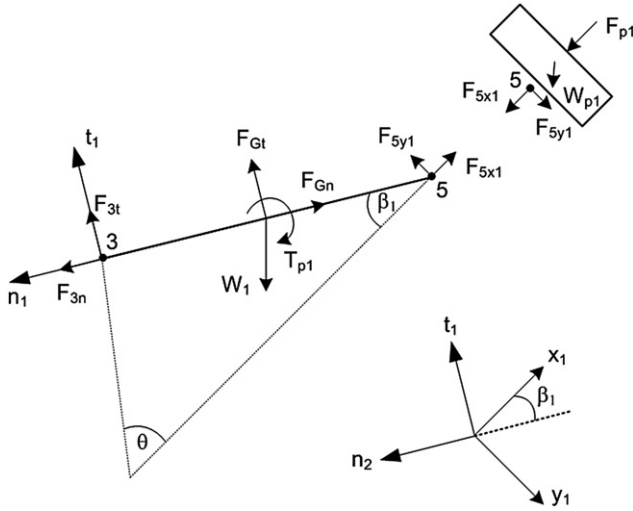


Fig. 2 – Dynamic diagram of expansion piston and rod.

From which F_{5x} , F_{3n} and F_{3t} are obtainable, other expressions are:

$$\begin{aligned} T_{p1} &= I_{2-3} \alpha_{2-3} \\ F_{p1} &= \pi r_{e1}^2 (P_{e1} - P_{ca}) \\ F_{5y} &= -F_{p1} - W_{p1} + m_{p1} a_{p1} \\ F_{Gn} &= \frac{1}{2} L m_{5-3} \omega_{5-3}^2 \\ F_{Gt} &= \frac{1}{2} L m_{5-3} \alpha_{5-3} \\ W_1 &= m_{5-3} \cdot g \\ W_{p1} &= m_{p1} \cdot g \end{aligned} \quad (24)$$

From all the equations in (24) it is worth saying that F_{p1} is the force that the working gas exerts over the piston, as a result from the pressure difference between the cylinder and the crankcase obtained from the thermodynamic model. Now, tangential force over the crankshaft can be calculated:

$$F_{\tan} = F_{3n} \sin(\beta_1 + \theta) + F_{3t} \cos(\beta_1 + \theta) \quad (25)$$

Torque exerted by the expansion system is:

$$T_1 = F_{\tan} \cdot R_{cig} \quad (26)$$

Force and torque diagram for the compression system is shown in Fig. 3. Torque expression is:

$$\begin{aligned} \sum T_3 &= T_{p2} - \frac{L}{2} G_{Gt} - L G_{4y} \sin \beta_2 + L G_{4x} \cos \beta_2 \\ &+ \frac{L}{2} W_1 \sin \left(3 \frac{\pi}{4} + \beta_2 \right) = 0 \end{aligned} \quad (27)$$

for n_2 direction, force balance is

$$\begin{aligned} \sum G_n &= -G_{Gn} + G_{3n} - G_{4y} \cos \beta_2 - G_{4x} \sin \beta_2 \\ &+ W_2 \cos \left(3 \frac{\pi}{4} + \beta_2 \right) = 0 \end{aligned} \quad (28)$$

and for t_2 direction,

$$\sum G_t = G_{Gt} + G_{3t} + G_{4y} \sin \beta_2 - G_{4x} \cos \beta_2 - W_2 \sin \left(3 \frac{\pi}{4} + \beta_2 \right) = 0 \quad (29)$$

from which F_{4x} , G_{3n} and G_{3t} are calculable. Other equations are:

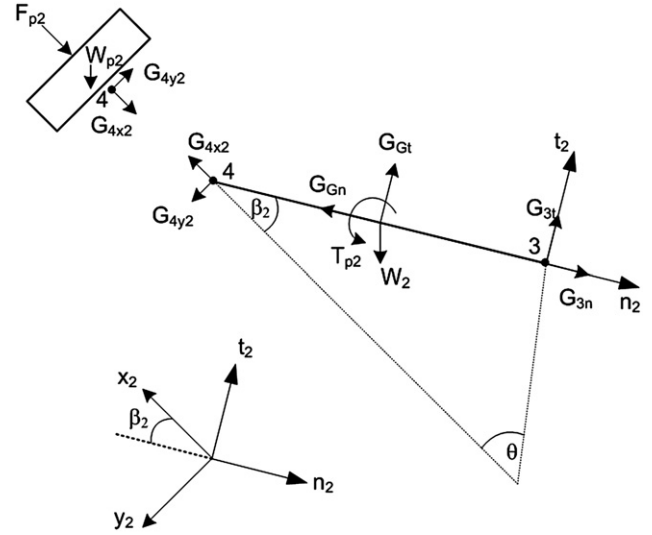


Fig. 3 – Dynamic diagram of compression piston and rod.

$$\begin{aligned} T_{p2} &= I_{4-3} \alpha_{4-3} \\ F_{p2} &= \pi r_{c2}^2 (P_{c2} - P_{ca}) \\ G_{4y} &= -F_{p2} - W_{p2} + m_{p2} a_{p2} \\ G_{Gn} &= \frac{1}{2} L m_{4-3} \omega_{4-3}^2 \\ G_{Gt} &= \frac{1}{2} L m_{4-3} \alpha_{4-3} \\ W_2 &= m_{4-3} \cdot g \\ W_{p2} &= m_{p2} \cdot g \end{aligned} \quad (30)$$

The tangential force exerted by the compression piston and rod over the crankshaft is:

$$G_{\tan} = G_{3n} \sin \left(\frac{\pi}{2} - \theta + \beta_2 \right) + G_{3t} \cos \left(\frac{\pi}{2} - \theta + \beta_2 \right) \quad (31)$$

Compression torque is:

$$T_2 = G_{\tan} \cdot R_{cig} \quad (32)$$

Power loss by friction is calculated from a simplified function obtained in [6]:

$$N_f = 3 \times 10^7 \cdot \left(\frac{30}{\pi} \omega \right)^{3.0154} \quad (33)$$

So the friction torque can be calculated as:

$$T_f = \frac{N_f}{\omega} = 3 \times 10^7 \cdot \left(\frac{30}{\pi} \right)^{3.0154} \cdot \omega^{2.0154} \quad (34)$$

The crankshaft torque is:

$$T_{cig} = I_{cig} \cdot \alpha \quad (35)$$

If I is the moment of inertia of the flywheel, then

$$I \frac{d\omega}{dt} = T_1 + T_2 + T_{cig} - T_f - T_q \quad (36)$$

Where T_q is external load torque, given by the connected mechanism. From this expression ω can be calculated at any time step.

Model flow chart of the computational process is shown in Fig. 4. Where:

$$w^{i+1} = w^i + \frac{I}{d\tau} (T_1 + T_2 + T_{cig} - T_f - T_q) \quad (37)$$

$$\theta^{i+1} = \theta^i + w^{i+1} d\tau \quad (38)$$

$$\alpha^{i+1} = (w^{i+1} - w^i) / d\tau \quad (39)$$

3. Prototype redesign and fabrication process

A previously built prototype published in [8] was redesigned and constructed in order to improve its performance. Several changes were made including the external heat supply device for tests, heater design, cooler design, regenerator, flywheel size, piston design, static and dynamic sealing mechanisms. Main specifications of the engine are shown in Table 1.

3.1. External heat supply for tests

First an electric oven was designed and calculated according to the power required to achieve the design temperatures of

Table 1 – Specification of engine main parameters.

Main specifications of the engine		
Power	428.22	W
Reference pressure	0.66	Mpa
Swept volume	439.707	cm ³
Frequency	13.83 (830)	1/s (rpm)
Beale number	0.11	
Phase angle	90°	
Working gas	Helium	

the engine by radiation (Fig. 5). A PID controller was installed for the manipulation of the power supplied, controlling the gas temperature inside the expansion exchanger. To achieve this, PT100 temperature sensors were incorporated inside the tubes of the expansion exchanger.

After performing the preliminary tests, it was noticed that the heat transfer was not the expected one and that there were dead spots of radiation. Therefore a gas burner was designed with the aim of achieving heat transfer mainly by convection. This alternative managed to maintain a more uniform temperature throughout the expansion exchanger. See Fig. 6.

3.2. Heater, cooler and regenerator sizing

Details of the design process which was made by a scaling methodology can be seen in [6]. The resulting specification of the exchangers is shown in Table 2.

3.3. Improvements to the heater

After construction and testing it was noted that there was a deficit of heat exchange area in the outside of expansion exchanger. To solve this problem fins were added to increase the exchange area as shown in Fig. 7. Fig. 6 shows the burner-expansion exchanger set with fins in operation.

3.4. Flywheel sizing

An important instance in improvements was to adopt a flywheel of greater inertia than the one used in [8], since it was

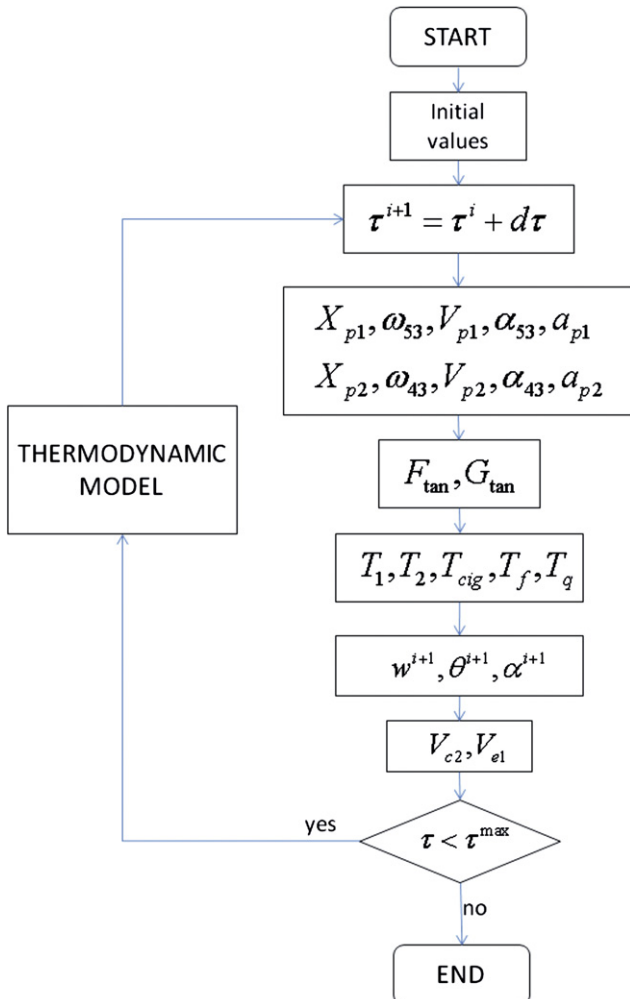


Fig. 4 – Model flow chart.

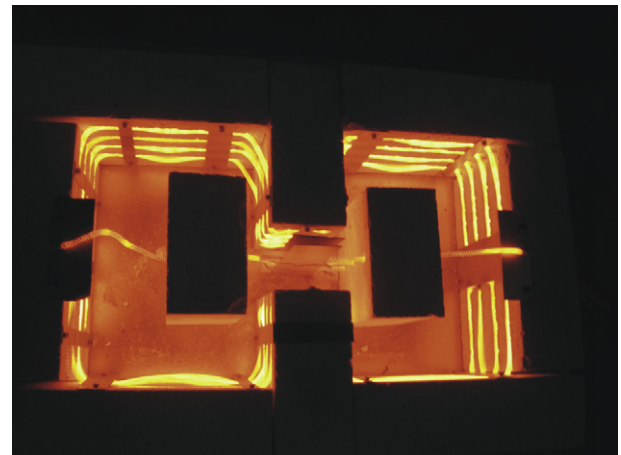


Fig. 5 – Electric oven used as external heat source in tests.



Fig. 6 – Expansion exchanger with burner incorporated inside the combustion chamber.

observed that the existing one did not store enough energy to overcome the work of compression and mechanical losses of a complete machine cycle.

The calculation described below verifies this. From this result, it was proceeded to find a flywheel on the market, whose moment of inertia was equal to or greater than the estimated one. After this, a new shaft design for the coupling of the engine to the electric starter was necessary. This included the selection of bearings, shaft coupling and the flywheel support design.

The flywheel of kinematic Stirling engines plays a very important role, which is to perform the compression work required and also to overcome the friction work loss in that part of the cycle where the machine is not delivering power. In many engines this is not taken into account and on certain occasions it can cause the engine to fail to keep functioning despite having a good thermodynamic performance. That is why in this work a specific equation to calculate the required moment of inertia of the flywheel is developed. The compression work, assuming it isothermal is:



Fig. 7 – Engine with new finned expansion exchanger.

$$W_c = \int_{V_1}^{V_2} P dV \quad (40)$$

Considering the gas as ideal, $P = \frac{mRT}{V}$

$$W_c = \int_{V_1}^{V_2} P dV = mRT \int_{V_1}^{V_2} \frac{1}{V} dV \quad (41)$$

$$W_c = mRT \ln(V_2/V_1) \quad (42)$$

In this case, taking into account a charge pressure of 6 bar the parameters are calculated in Table 3.

The kinetic energy K stored in the flywheel is:

$$K = \frac{1}{2} I \omega^2 \quad (43)$$

And,

$$W_c = -\Delta K \quad (44)$$

So the moment of inertia can be calculated as:

$$I = \frac{2\Delta K}{(\omega_2^2 - \omega_1^2)} = -\frac{2W_c}{(\omega_2^2 - \omega_1^2)} \quad (45)$$

The required moment of inertia will be the one that allows an acceptable range of angular velocity variation. If the acceptable relative angular velocity variation is defined as:

$$\Delta\omega_r = \frac{\Delta\omega}{\omega} = \frac{\omega_2 - \omega_1}{\omega_1} \quad (46)$$

Table 2 – Specification of engine exchangers.

Compression exchanger	
Number of tubes	43
Length of tubes	181 mm
Int. diameter of tubes	3.45 mm
Expansion exchanger	
Number of tubes	10
Length of tubes	446.2 mm
Int. diameter of tubes	4.35 mm
Regenerator	
Length	54.26 mm
Hydraulic radius	0.117 mm
Free flow area	2474 mm ²
Wire diameter of mesh	0.21 mm
Mesh number	1.874
Number of layers	128
Mesh surface	3558 mm ²
Mesh layers diameter	67.57 mm

Table 3 – Gas characteristics and resulting compression work.

m	7,82E-04 kg
R	2078.6 J/kg K
T	300 K
V ₂	8.1325e-004 m ³
V ₁	3.8021e-004 m ³
W _c	–371,00J

Where ω is the desired average angular velocity in rad/s. Now ω_2 can be replaced in equation (45) with its expression from equation (46). Operating and making $\omega = \omega_1$ the equation for the required moment of inertia of the flywheel is:

$$I = \frac{-2W_c}{\omega^2(\Delta\omega_r^2 + 2\Delta\omega_r)} \quad (47)$$

With the engine data and considering the value of the necessary moment of inertia is 0.91 kg.m².

So far the friction work W_f was not considered. It can be calculated from the friction power N_f as:

$$W_f = -\frac{N_f}{\omega} d\theta \quad (48)$$

Where the value of θ to consider is radians, considering compression occupies about half of the cycle approximately. Then,

$$W_f \approx -\frac{\pi N_f}{\omega} \quad (49)$$

The calculation of the moment of inertia including the friction work is then:

$$I = \frac{-2\left(W_c + \frac{\pi N_f}{\omega}\right)}{\omega^2(\Delta\omega_r^2 + 2\Delta\omega_r)} \quad (50)$$

In this case the calculated friction power is 200 W, so the needed moment of inertia is 0.93 kg.m². It is thus noted that the friction work does not significantly influence the calculation.

3.5. Redesign of pistons and seals

The pistons of the previous engine [8] were the original ones of the base machine used (an air compressor). Improvement in the design of the expansion piston was decided in order to increase the distance between the heat source and piston rings, since high temperatures can affect the sealing materials.

In Fig. 8 the stainless steel extension drawing of expansion piston can be seen. Inside it, horizontal baffles were placed in

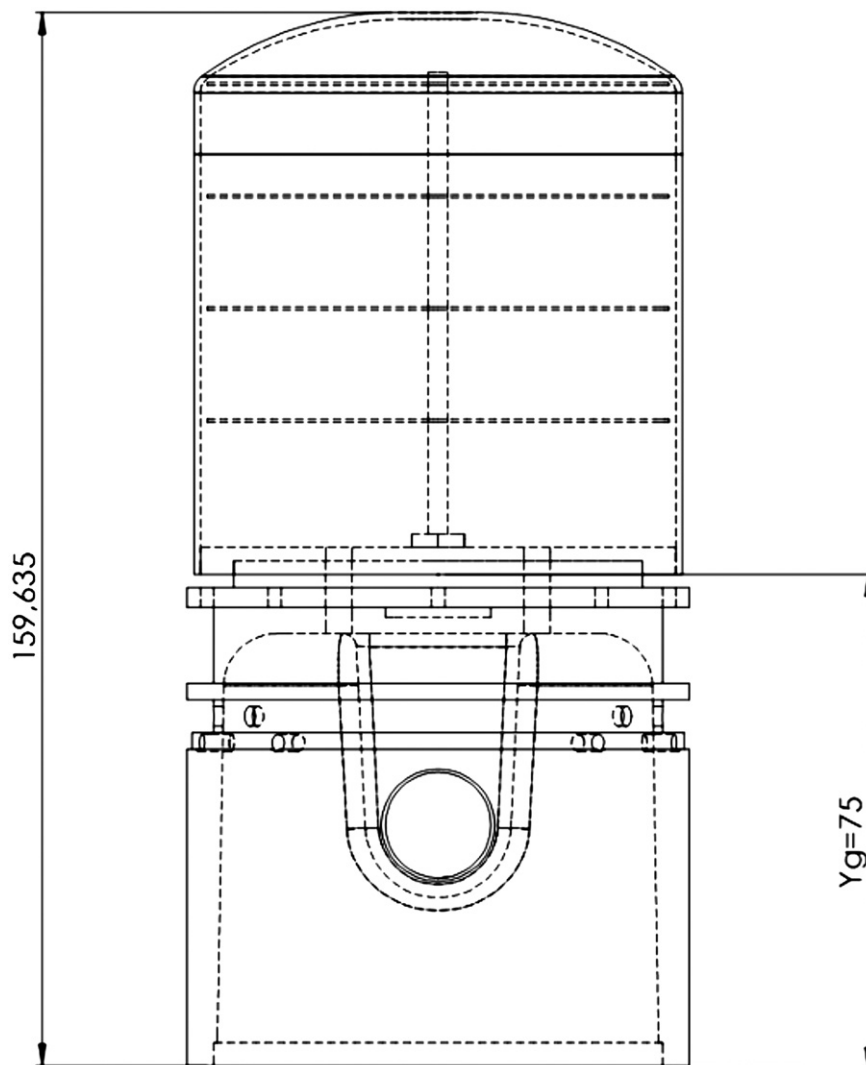


Fig. 8 – Expansion piston. Design detail.



Fig. 9 – Reinforced PTFE rings.

order to prevent the heat convection and radiation between the hot part (top) and the cold one (bottom).

As regards the piston seal, in the earlier constructed device [8] it had been assembled with the original rings of the air compressor that are made of steel. They allowed a great amount of gas passing from the cylinders to the crankcase during compression and expansion.

To reduce such losses of gas to the crankcase, reinforced PTFE rings backed up by inside o-rings were designed. Fig. 9 shows the ring that works as a dynamic seal. Likewise, a PTFE guide ring was added for the piston to move smoother and avoid erosion. Fig. 10 shows the overall assembly of the expansion piston of the thermal machine.

3.6. Tests

Tests were performed for the equipment working as a refrigerator in order to make it better comparable with previous tests. During the test the temperature evolution in the expansion exchanger's wall was measured as the engine was moved by an auxiliary electric motor.

This test was conducted for 9 min. The following graph shows this evolution compared with that obtained in [8] for



Fig. 10 – View in detail of the piston rings assembled.

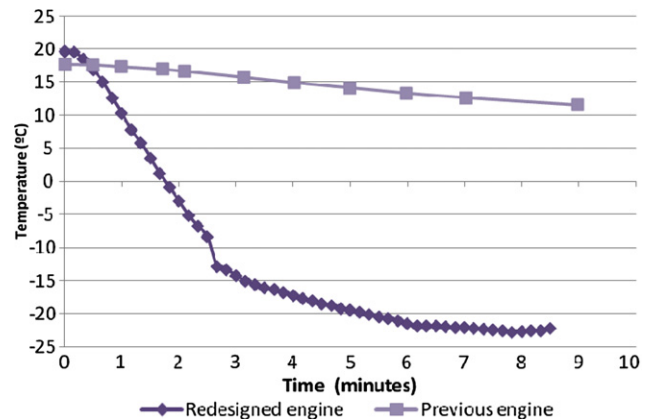


Fig. 11 – Evolution graphic comparing the temperature of the expansion exchanger tubes in the new and old design respectively.

the previous design of the engine. It should be mentioned that they are not directly comparable because the one from [2] was performed at 7 bar and 200rpm while the current one was held at 2.4 bar and 830 rpm. Furthermore, 1 L of water was placed in a flexible container in contact with the exchanger's tubes. It can be seen in the graph of Fig. 11 that the temperature reached -23°C .

4. Results

As regards the model developed, Fig. 12 shows angular velocity variation in time, for different flywheel moments of inertia ($6\text{ kg}\cdot\text{m}^2$; $0.6\text{ kg}\cdot\text{m}^2$; $0.06\text{ kg}\cdot\text{m}^2$), all of them for an initial angular velocity of 850 rpm. If less inertia is used, the angular velocity decreases rapidly, meaning that the engine does not work for this pair of I and ω .

For greater moments of inertia, angular velocity establishes in a longer time, but its amplitude is lower, representing a better running. As the steady-state regime is reached, the average angular velocity approaches the same value, which is near 1900rpm, for all values of I used in Fig. 12.

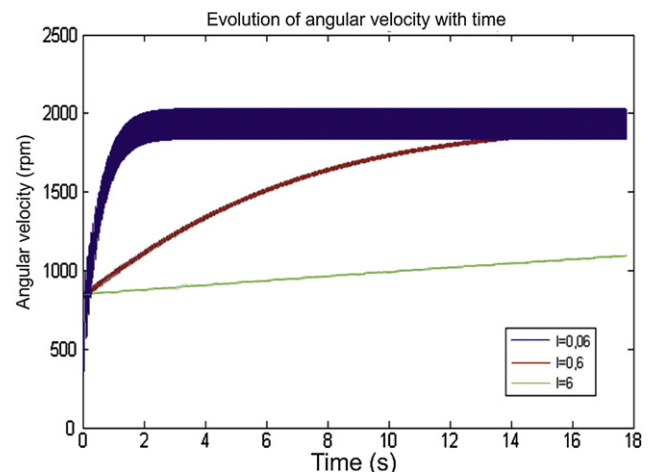


Fig. 12 – Angular velocity as a function of time, for different flywheel moments of inertia.

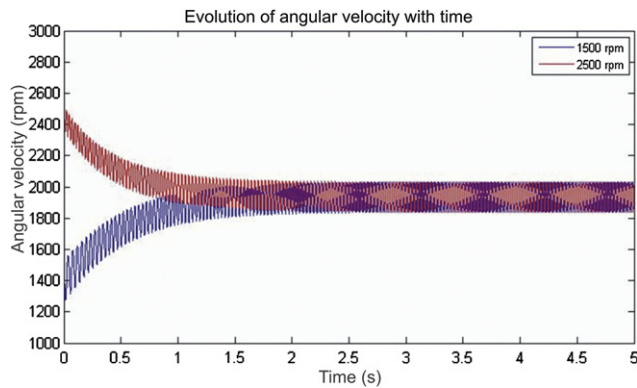


Fig. 13 – Angular velocity in time, for different initial velocities.

Fig. 13 shows angular velocity variation in time, for a flywheel moment of inertia of 0.06 kg·m², but different initial velocities (1500rpm and 2500rpm).

For initial rotational velocities under 850rpm, velocity function decreases fast. Above that initial condition, regardless of its value, ω establishes on 1900 rpm. Amplitudes, after transient, are both equal.

5. Conclusions

The elaboration process of the model allows a better understanding of the system as a whole. The thermodynamic model used is a basic isothermal one, with results compatible with those of Schmidt analysis. Its simplicity is used to determine if the mechanical model is running as it should. Once this is checked, an improvement of thermodynamic model should be done, in order to approach its results to the experimental ones, and make more reliable predictions.

Next version of this model should solve the influence of variables such as piston, rod and crankshaft weight over performance, rotational velocity and power, in connection with pressure loss, heat loss and regenerator performance.

With reference to the prototype, as mentioned above, the choice of a gas burner as the heat source initially gives good results, but it is to consider the possibility of increasing the airflow to improve heat transfer by convection to the finned

tubes. The addition of fins to the expansion exchanger tubes increased the heat transfer area, producing good results, as it turned out that the temperature was more distributed in the tubes as shown in Fig. 6 and also an increase in the rate of heat transfer.

The developed equation for the flywheel allows a quick and simple sizing for future applications.

The redesign work on pistons and seals showed very good results, as it was possible to measure a pressure difference between the thermodynamic circuit and the crankcase, which demonstrates that the new reinforced PTFE rings are working as expected. Measurements of mechanical friction should be carried out. As for static seals, their performances were very satisfactory since working fluid losses were not detected.

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